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THE EFFECT OF A WOODRUFF BEARING IN ROTARY COMPRESSOR

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ABSTRACT

This paper deals with the efficiency improvement of a rolling piston type rotary compressor through numerical and experimental investigation of an eccentric bearing.

We numerically and experimentally analyzed mechanical loss of an eccentric bearing to improve mechanical efficiency of a rotary compressor. The numerical results show that an eccentric bearing with cut off frictional surface gives higher mechanical efficiency than that of a normal eccentric bearing. The experimental results correspond to the numerical results. We named this eccentric bearing with cut off frictional surface as a Woodruff bearing which is shown in Fig.1.

In addition, to confirm the reliability of a Woodruff bearing, we analyzed in detail the oil film pressure under conditions of an optimum shape and dimensions by applying " Finite Element Method "(FEM).

This paper concludes that a Woodruff bearing with an optimum shape is effective for obtaining higher efficiency.

INTRODUCTION

In recent years, the requirement for high EER compressor is growing. In order to increase EER, it is essential to reduce the bearing loss. For the implementation, it is effective to reduce the bearing width or bearing diameter. However, the excessive reduction results in decrease of bearing load capacity. In order to reduce bearing loss without reducing the load capacity,if the bearing load direction is constant, it is effective to cut off the frictional surface of the anti-load side. A partial bearing is based on this concept. The bearing load direction of a rotary compressor rotates. However, the direction of high load does not rotate but remain within a constant range. Therefore, it is effective to apply a partial bearing concept for reducing the bearing loss of a rotary compressor.

We obtained an optimum shape and dimensions of a Woodruff bearing by numerical analysis of mechanical loss under the actual load.

NOMENCLATURE

Me = frictional moment at eccentric bearing
Rr = piston radius
Lc = cylinder height
Le = eccentric bearing length
Rc = cylinder radius
Re = eccentric bearing radius
C = radial clearance between eccentric bearing and piston
Ps = pressure at suction process
Pd = pressure at discharge process
F = eccentric bearing load
ω = angular velocity of shaft
ωp = angular velocity of piston
ε = attitude of bearing
δ = attitude angle of bearing
μ = viscosity of lubricating oil
ηd = product of mechanical efficiency and indicated efficiency
θf = directional angle of bearing load
θ = coordinate of bearing angle
Z = coordinate of bearing length
ψ = eccentric bearing load direction
$\cdot$ = time differential

NUMERICAL ANALYSIS

We developed an engineering program to obtain optimum shape which gives a higher mechanical efficiency. The engineering program analyzes mechanical loss and minimum oil film thickness of an eccentric bearing under the conditions of various bearing shapes and dimensions.

The detail of the engineering program is as follows :
An eccentric bearing consists of a crankshaft and a piston.[Fig.2] The analysis of the bearing is based on the assumption that a piston does not rotate and a shaft is not eccentric.[Fig.3] The definitions of elements which are influential in bearing efficiency are :

1) Frictional surface where cut off area does not exist :
$W = \gamma 2 - \gamma 1 \ (= \beta 2 - \beta 1)$ [deg]
2) Frictional surface where cut off area exists :
$V = 360^{\circ} - (\gamma 2 - \gamma 1)$ [deg]
3) Direction of frictional surface where cut off area does not exist :
$\phi = (\gamma 1 + \gamma 2)/2$ [deg]
4) length of frictional surface where cut off area does not exist :
Le [mm] (= Eccentric bearing length)
5) length of frictional surface where cut off area exists :
Lw [mm]

Where $\gamma 1$ and $\gamma 2$ are the angles from eccentric direction to boundary between frictional surface with and without cut off area [Fig. 4].

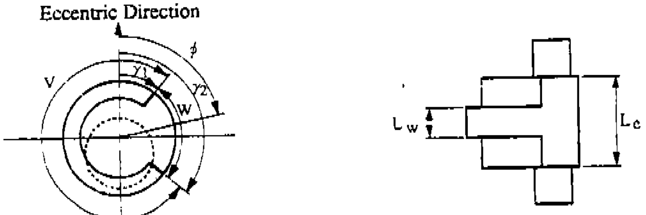

Fig.4 The Definition of Elements

The following equation gives frictional moment Me for a normal eccentric bearing,

$$Me = \frac{2 \cdot \pi \cdot \mu (\omega - \omega p) Le \cdot Re^3}{C \sqrt{1-\varepsilon^2}} - \frac{1}{2} C \cdot \varepsilon \cdot F \cdot \sin \delta \quad \text{.................} \quad (1)$$

where ε and δ are obtained from the oil film pressure analysis. When applying equation (1) to a Woodruff bearing, the length of frictional surface where cut off area exists needs to be changed from Le to Lw. Therefore, the following equation gives frictional moment Me for a Woodruff bearing,

$$Me = \frac{2 \cdot \mu (\omega - \omega p) R e^3}{C \sqrt{1-\varepsilon^2}} \left[Lw \{ A(\beta 2) - A(\beta 1) \} + Le \{ A(\beta 1 + 2\pi) - A(\beta 2) \} \right] - \frac{1}{2} C \cdot \varepsilon \cdot F \cdot \sin \delta \quad \text{.................} \quad (2)$$

where

$$A(\theta) = \tan^{-1}\left(\sqrt{\frac{1-\varepsilon}{1+\varepsilon}} \tan \frac{\theta}{2} \right)$$

and where $\beta_1 = \gamma_1 - \delta - \psi$、 $\beta_2 = \gamma_2 - \delta - \psi$

For obtaining oil film pressure, it is also necessary to change Le to Lw, when applying equation for a full journal bearing.

$$P(\theta, Z) = \frac{6 \cdot \mu \cdot Z \cdot (Le - Z)}{C^2 (1 + \varepsilon \cdot \cos \theta)^3} \left[\frac{\varepsilon}{2} \{ \omega + \omega p - 2(\dot{\theta}_f + \dot{\delta}) \} \cdot \sin \theta - \varepsilon \cdot \cos \theta \right] \quad (3)$$

The Gümbel boundary condition is applied to this equation. The study referred to for this analysis is in [1–2]. The following tables show the dimensions of a compressor and the conditions for analysis.

Table 1 Dimensions of Compressor

R_c [mm]	22.0
L_c [mm]	25.0
R_r [mm]	17.8
R_e [mm]	13.35
L_e [mm]	17.0

Table 2 Analysis Conditions

P_s / P_d [MPa]	0.58／2.17
speed [rpm]	5400
volume [cc / rev]	13.2

NUMERICAL RESULT

Fig 5-10 show the numerical results of the mechanical loss and the minimum oil film thickness of a Woodruff bearing under the various conditions of ϕ, W and Lw.

How ϕ affects efficiency

The results show that at ϕ = Approx. 90° the mechanical loss is minimized and the minimum oil film thickness is maximized.[Fig.5 and 8] The cause of these results can be explained as follows :

During the period of low bearing load (at small crank angle), the load is on the side of cut off frictional surface.

During the period of high bearing load(at large crank angle), the load is on the side of cut off frictional surface.

How W affects efficiency

At $\phi = 90^\circ$, the mechanical loss decreases by reducing W and the minimum oil film thickness slightly decreases until W = Approx. 120° , but beyond this point the minimum oil film thickness sharply drops by reducing W.[Fig.6 and 9] Therefore, W = 120° should be selected as optimum W.

How Lw affects efficiency

At $\phi = 90^\circ$, as Lw decreases, the mechanical loss decreases with a small reduction of oil film thickness.[Fig.7 and 10]

The mechanical loss is minimized under the conditions of $\phi = 90^\circ$,W = 120° and lower Lw with a small reduction of minimum oil thickness.

EXPERIMENT

We conducted experiment of a compressor currently in production by varying bearing shapes of a crankshaft. Table 3 shows the shapes of eccentric bearings used for the experiment.

Table 3 Dimensions of Eccentric Bearing

	1	2	3	4	5	6	7
ϕ [deg]	–	0	90	180	270	90	90
W [deg]	360	120	←	–	←	–	180
L_W [mm]	17	5	←	←	–	10	5

The tolerance of shaft diameter, eccentric diameter and eccentricity of a crank shaft is within ±2 μm. An automatic controlled calorimeter was used for the measurement of compressor performances.

EXPERIMENTAL RESULT

Fig.11-13 show the experimental results in comparison with the numerical results. These results show the relation of ϕ , W and Lw and the product of mechanical efficiency and indicated efficiency. The experimental results correspond to the numerical results. Under the conditions of $\phi = 90^\circ$,W = 120° and Lw = 5mm, the efficiency increases by 0.7 %. The results show that a Woodruff bearing gives better efficiency than that of a normal eccentric bearing and the engineering program is applicable to numerical analysis for obtaining the optimum shape of a Woodruff bearing.

OIL FILM PRESSURE ANALYSIS BY FEM

We analyzed in detail the oil film pressure on the bearing of an optimum shape ($\phi = 90^\circ$,W = 120° , Lw = 5mm) by applying FEM. Fig.14 shows the oil film pressure at different crank angles. The results show that the oil film pressure is small at the boundary region between cut off and non cut off area where the bearing load is comparatively small. The results also show that the oil film pressure is large on the frictional surface without cut off area where the bearing load capacity and the load are large. Therefore, the reliability can be maintained. The partial bearing concept is applicable to the Woodruff bearing.

CONCLUSION

Numerical and experimental analysis show that a Woodruff bearing of a rotary compressor gives higher efficiency (the product of mechanical efficiency and indicated efficiency) than that of a normal eccentric bearing.

The results of the newly developed engineering program correspond with the experimental results. Therefore, it is applicable to the engineering of an eccentric bearing used for a rotary compressor. From this program, mechanical loss and minimum oil film thickness of an eccentric bearing are obtainable , and accordingly the dimensions and the shape are determined.

REFERENCE

[1] Yanagisawa, T. , et al.,"Motion Analysis Of Rolling Piston In Rotary Compressor", Proc. Purdue Compressor Tech. Conf. , 1982

[2] Nakagawa, E., et al.,"A Calculation Method Of Characteristic Performance Of Journal Bearings Under Dynamic Loading",(in Japanese), Lubrication, Japan, Vol.15 No.7, 1970, pp.385-390

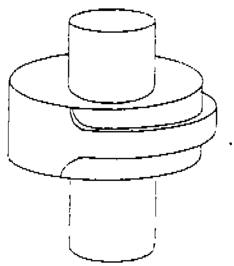

Fig.1 Scheme of Woodruff Bearing

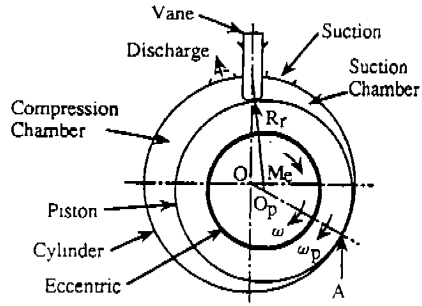

Fig 2 Scheme of Rotary Compressor

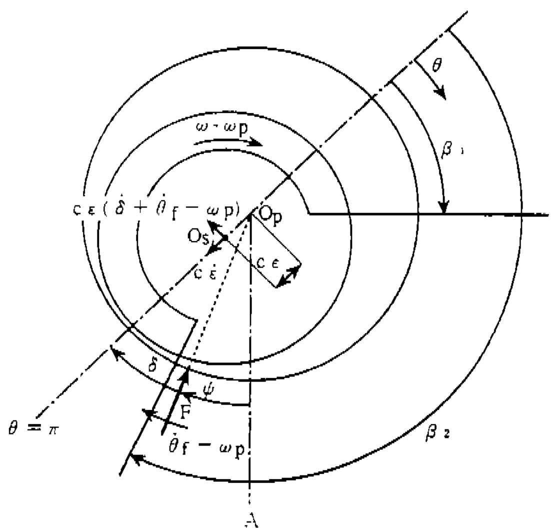

Fig. 3 Eccentric Bearing Model

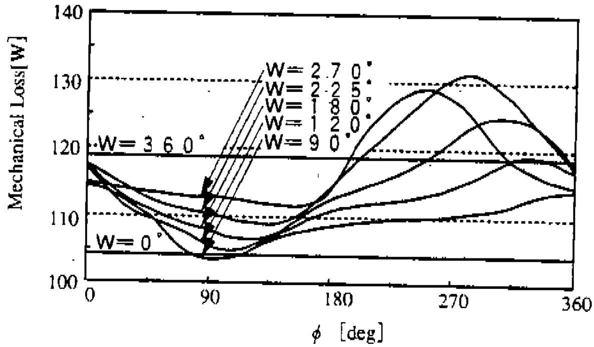

Fig.5 Relation Of ϕ And Mechanical Loss By Varying W Under Condition Of Lw = 5 mm (Calculated)

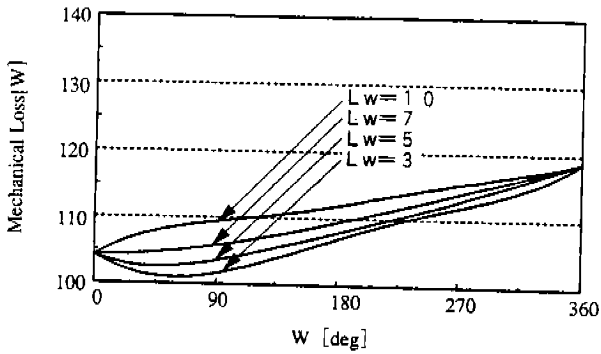

Fig.6 Relation Of W And Mechanical Loss By Varying Lw Under Condition Of $\phi = 90°$ (Calculated)

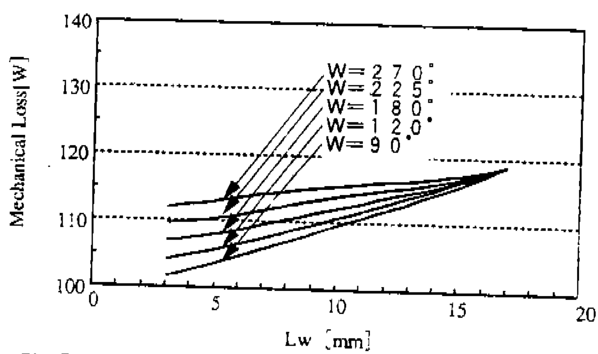

Fig. 7 Relation Of Lw And Mechanical Loss By Varying W Under Condition Of $\phi = 90°$ (Calculated)

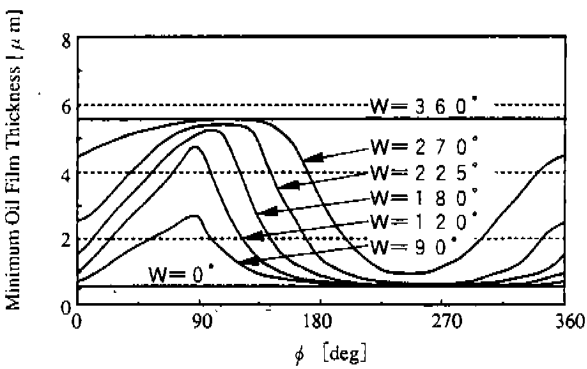

Fig.8 Relation Of ϕ And Minimum Oil Film Thickness By Varying W Under Condition Of Lw= 5 mm (Calculated)

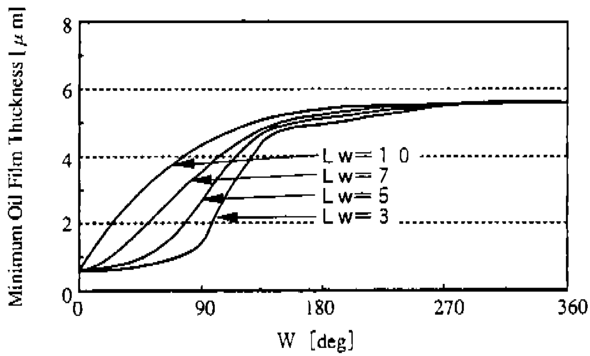

Fig 9 Relation Of W And Minimum Oil Film Thickness By Varying Lw Under Condition Of ϕ = 9 0° (Calculated)

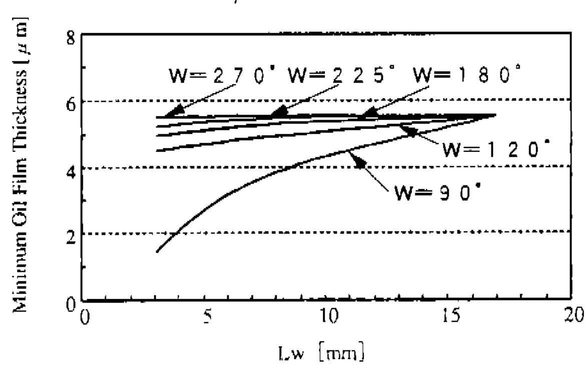

Fig 10 Relation Of Lw And Minimum Oil Film Thickness By Varying W Under Condition Of ϕ = 9 0° (Calculated)

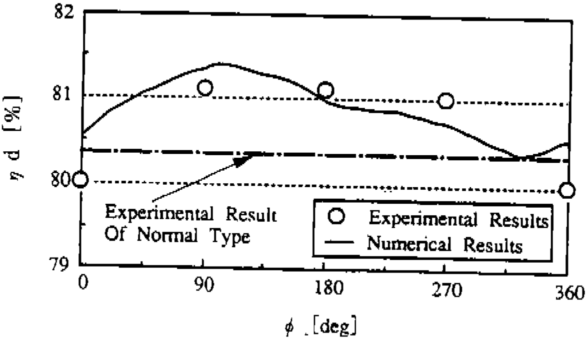

Fig.11 Product Of Mechanical Efficiency And Indicated Efficiency (η d) Comparison With Experimental And Numerical Results By Varying ϕ Under conditions of W = 1 2 0 ° And L w = 5 mm

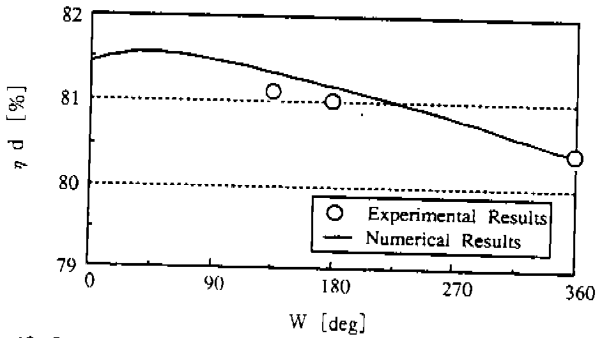

Fig.12 Product Of Mechanical Efficiency And Indicated Efficiency (η d) Comparison With Experimental And Numerical Results By Varying W Under conditions of ϕ = 9 0 ° And L w = 5 mm

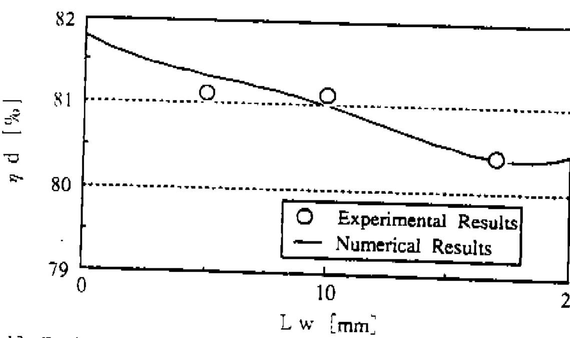

Fig.13 Product Of Mechanical Efficiency And Indicated Efficiency (η d) Comparison With Experimental And Numerical Results By Varying L w Under conditions of ϕ = 9 0 ° And W = 1 2 0 °

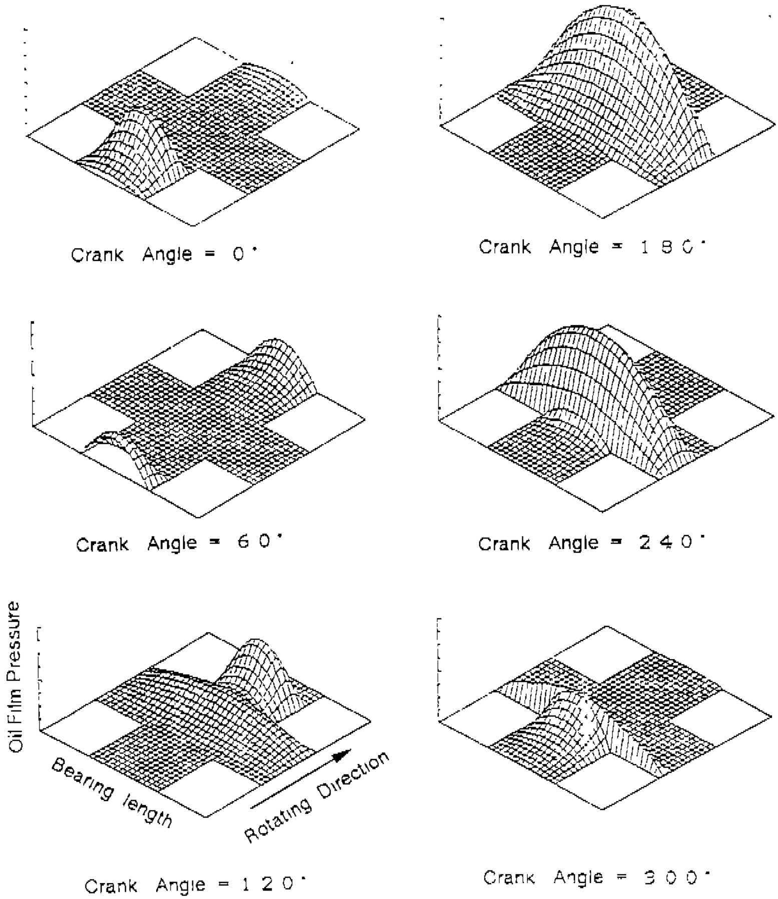

Fig. 14 Oil Film Pressure Distribution On Eccentric Bearing With Optimum Shape